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EVALUATION OF CRANKSHAFT DESIGNS PROPOSED FOR THE DRY AND WET HELIUM EXPANSION ENGINES

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FOR

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I N D E X

1. INTRODUCTION:

This report summarizes a design review of alternate crankshaft designs and the load carrying ability of tandem bearings proposed for both the wet and dry helium expansion engines originally reviewed and reported in CCI Report No. 390-110.

2. SCOPE:

Specific questions to be answered when evaluating the proposed designs were as follows:

2.1 Ascertain the load carrying ability of the tandem bearing design crankshaft.

2.2 Evaluate the design of the enclosed Model #2 crankshaft from considerations of strength and reliability.

2.3 Suggest an alternate design to Model #2.

2.4 Using the above information, present a design for a crankshaft with a 3/8 in. throw for use on a wet engine in satellite refrigerator service.

The list of drawings used for this evaluation was as follows:

Dwg. No.	Rev.	Title
1820-MD-111504	B	Gas Expansion Engine Crankshaft
1820-MB-111503	B	Gas Expansion Engine Crank Bushing
1820-MD-111791	A	Gas Engine Crankshaft Long Half
1820-MC-111774	A	Gas Engine Crankshaft Short Half
1820-MD-111804		Liquid Helium Engine Crankshaft (Gas Engine Conversion)
1820-MB-111805	-	Liquid Helium Engine Crank Bushing (Gas Engine Conversion)

3. <u>LOAD CARRYING ABILITY OF TANDEM BEARINGS</u>:

Theoretically, the radial load applied to the two tandem mounted crank bearings through the connecting rod should be equally shared by each bearing if the rocker arm and connecting rod linkage is properly aligned. In actual practice, however, slight tolerances or misalignment can create an out-of-balance situation, and a practical approach to compensate for this unknown load distribution is to reduce the Basic Dynamic Capability of the bearings by a factor of 80%.

Revised calculations using 80% of the bearings' rated BDC are as follows:

$$\text{Piston Force} = 20 \text{ ata} \times 14.7 \times 3.187^2 \times \frac{\pi}{4}$$

$$= 2{,}346 \text{ lbs}$$

$$\text{Connecting Rod Force} = 2{,}346 \times \frac{2}{1}$$

$$= 4{,}692 \text{ lbs}$$

Load per bearing (2 Bearings) =

$$\frac{4692}{2} = 2{,}346 \text{ lbs}$$

3.1 <u>Present Bearing</u>:

McGill Bearing #SB-22211

Rated BDC = 21,100 lbs

80% BDC = 16,880 lbs

At 500 RPM,

Equivalent RPM = 561 RMP F_S = 2.33

$$\text{Life Factor } (F_L) = \frac{16880}{2346 \times 2.33 \times 2} = 1.54$$

(B-10) Life = 2,100 hrs at 500 RPM

It is recommended that the SB-22212 bearings be used if possible; however, other factors must also be considered. The bearing cost itself is relatively inexpensive. If routine maintenance is already scheduled at approximately 2,500 hrs, it may be practical to inspect and/or change bearings at that time.

4. EVALUATION OF MODEL #2 GAS ENGINE CRANKSHAFT:

Drawings 1820-MD-111791 and 1820-MC-111774 were reviewed to evaluate the design from considerations of strength and reliability.

4.1 The bending stress in the short half shaft extension is:

$$S = \frac{MC}{I}$$

Where $M = 2{,}346$ lbs x 1 in.

$C = 0.5$ in.

$I = .049 \times d^4 = .049 \text{ in.}^4$

$$S = \frac{2346 \times 1 \times .5}{.049} = 23{,}939 \text{ psi}$$

This stress is rather high and, because a slight redesign can provide a 1-1/4 in. diameter shaft extension, the stress level can be greatly reduced.

$$S = \frac{MC}{I}$$

Where $M = 2{,}346$ lbs x 1 in.

$C = .625$ in.

$I = .049 \times 1.25^4 = .1$

$$S = \frac{2346 \times 1 \times .625}{.12} = 12{,}220 \text{ psi}$$

Increasing the shaft extension to 1-1/4 in. diameter almost halves the bending stresses in the shaft.

7. EVALUATION OF WET ENGINE CRANKSHAFT (MODEL #1):

Drawings 1820-MD-111804 and 1820-ME-111805 were reviewed. It appears from the diameters specified that McGill bearings SB-22211 will be used, both on the crank and on the end supports. Therefore, the bearing analysis was based upon these bearings.

7.1 Bearing Life:

Piston Force $= 20 \text{ ata} \times 14.7 \times 1.5^2 \times \frac{\pi}{4}$

$= 520$ lbs

Connecting Rod Force $= 520 \times \frac{2}{1} = 1{,}040$ lbs

Load Per Bearing (2 Bearings):

$\frac{1040}{2} = 520$ lbs

Rated BDC = 21,100 lbs

80% BDC = 16,880 lbs

At 500 RPM

Equivalent RPM = 561 RPM $F_s = 2.33$

Life Factor $(F_L) = \frac{16{,}880}{520 \times 2.33 \times 2} = 6.96$

(B-10) Life > 100,000 hrs

7.2 Shaft Stress:

$$S = \frac{520 \times 1'' \times .625}{.12} = 2{,}710 \text{ psi}$$

This is a very low stress level which is fine.

7.3 Bearing Shoulders:

As in the previous cases, the crankshaft and bushing must both be provided with shoulders so that bearing outer race does not rub against a stationary surface or vice-versa.

Provide a 2-1/2 in. diameter x 1/32 in. high shoulder for both the end bearings and the crank bearings on the crankshaft.

Provide a 1-3/4 in. diameter x 1/32 in. high shoulder on the crank bushing.

7.4 Add a note or tolerance to the bushing drawing 1820-MB-111805 that specifies concentricity of the 1.250 in. diameter to the 2.1654 in. diameter within .0005 in. TIR.

7.5 The exact purpose of all the tapped holes is unknown. It is thought that perhaps they are for jacking bolts to assist in bearing removal.

Some thought might be given to locating them closer to the inner race; however, the addition of the shoulders to hold the bearing away from the main plate provides clearances where tapered wedges could be employed.

7.6 The value of the #8-32 screw connecting the bushing to the crankshaft is questionable. The screw is too small to take any torque which will be taken by the keyway anyway, and the two pieces are adequately clamped by the 7/8-14 nut on the end of the crankshaft. Unless there is some other use for this tapped hole, not readily apparent, it is advised to delete the hole.